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Improvements in or relating to swirling fluid flows and jets

The present invention relates to improvements in or relating to swirling fluid flows and jets. In particular it concerns apparatus for controlled expansion (or contraction) of a swirling fluid flow, and a means for creating an annular swirling fluid flow, which, it will be shown, is particularly amenable to expansion.

In general terms, a swirling fluid flow (destined to become a jet) is defined as any fluid with components of longitudinal (axial) and tangential (swirl) velocity that travels through, and subsequently emerges from, a circular duct into an unconfined ambient. An ambient is any like fluid, to that comprises the jet. An annular swirling fluid flow is here defined as one in which the velocity maxima of the longitudinal and tangential components are approximately coincident, and offset from the axis of the flow.

In general, the purpose of expanding a swirling fluid flow to form a jet, is in order to exploit the unusual topology (coherent flow structures) associated with an expanded swirling jet, and the varying flow velocity and pressure regimes within such a jet. Specifically, and in relation to the present invention, the purpose may be seen to be:

- To take advantage of the different ways in which a moderately expanded swirling jet interacts with the ambient fluid - for enhanced-thrust vessel propulsion.
- To take advantage of the unique flow regime associated with a highly expanded swirling jet - for the purposes of combustion and spraying.
- To utilise the impingement behaviour of a highly expanded swirling jet - for the purposes of seabed excavation and substrate cleaning.
- To take advantage of the enhanced rate of cooling and mixing (of a swirling gas flow) when a highly swirled flow is efficiently expanded - e.g. for refrigeration purposes. Note that for the same purpose, using the flared nozzle in reverse to cause flow contraction may have advantages in terms of increased-efficiency of thermal separation.

Typically, the fluid is water. It may also be gas (air) if the expansion/contraction process takes place substantially at atmospheric pressure.

A conventional (i.e. straight-sided) conical diffuser is a device, whose primary function is to induce a uniform cross-sectional area expansion of a (non-swirling) unidirectional fluid stream, thereby achieving a reduction in velocity and an increase in static pressure. However, because such expansion is forced, against the natural reluctance of non-swirling fluid flows to undergo widthways expansion, conical diffusers are, of necessity, very long relative to their width (cone angle typically less than10°). (See, for example Figure 7.10 of *Massey, B. Mechanics of Fluids. 7th Edition, 1998, Stanley Thornes (Publishers) Ltd.*). This is because, as the cone angle increases above about 12°, fluid begins to separate from the wall and conventional conical diffusers, thereafter, rapidly become very inefficient. Consequently, efficient flow expansion with conventional conical diffusers comes at a price in terms of

diffuser size (length) and weight, and cost of diffuser fabrication. Additionally, there are no beneficial characteristics, in terms of unusual jet behaviour, associated with an expanded unidirectional (non-swirling) flow.

In a first aspect of the present invention, there is provided a swirling fluid flow expansion apparatus comprising a flared nozzle having an inlet for operative connection to a swirling fluid flow, an outlet and an intermediate body having an elliptical inner profile with particular dimensional characteristics. These dimensional characteristics are dependent on the ratio of longitudinal and swirl velocity of the inlet flow (as defined by the Swirl Number), and on the desired characteristics of the outlet flow or jet.

Preferably, the inner profile of the nozzle forms a continuous smooth curve (as indicated in Figure 1).

In a second aspect, a swirling fluid flow generator is provided. In one arrangement the generator comprises an inlet duct for operative connection to a pressurised fluid source, an outlet duct coaxial with the inlet duct and an intermediate generator body comprising at least one aperture defining a fluid flow path between the inlet duct and outlet duct wherein the fluid flow path has an axis orthogonal to that of the outlet duct.

Preferably, the fluid flow path is defined by a plurality of apertures in the generator body.

Preferably, the outlet duct is circular in cross-section and each aperture is disposed tangentially to the outlet duct.

Preferably, four apertures are provided arranged in two axially spaced pairs; the apertures of each pair being co-planar and oppositely oriented.

Preferably, the four apertures are rotationally spaced at 90° to each other.

In a second arrangement of the second aspect the swirling fluid flow generator comprises an impeller mounted within a duct having an inlet and an outlet defining a fluid flow path; characterised in that the impeller includes an impeller body having a central axial longitudinal flow blocking portion, that extends out in the form of star-shaped arms and swirling flow inducing elements (blades) attached to, or formed with, the arms of the star.

Suitably, the impeller is formed from sheet metal.

Preferably, the central axial longitudinal flow-blocking portion is generally circular and the star-shaped arms extend out in symmetrical fashion.

Suitably, wherein the swirling flow inducing elements (blades) are attached to the star along a tangential line.

Preferably, the impeller blade is inclined at an angle of 30° to 45° from the plane of the impeller; more preferably about 30°.

In a third aspect, the present invention provides a swirling fluid flow apparatus comprising a combination of swirling fluid flow generator and swirling fluid flow expansion apparatus, as described above.

According to the present invention there is provided, as a first component, a device for efficient expansion of swirling fluid flows, namely a flared nozzle; placed over the duct outlet (or orifice) supplying the swirling flow, or as a transition between two ducts of differing diameters. The flared nozzle is designed to work with all types of swirling flow. It has an inlet and an outlet, and an intermediate body having pre-determined dimensional characteristics, which are described in detail herein.

The flared nozzle can be likened, functionally, to a conventional conical diffuser. It will be shown later that there is also a generic link, in that (in the limit) as the swirl component of the flow approaches zero, the geometry of the flared nozzle converges to that of a conventional conical diffuser. However, for the purposes of the herein described invention and its applications (with moderate to highly swirled flows), the flared nozzle can be seen as differing physically, in that the internal angle increases lengthways from 0° to 180°, and the length-to-width ratio is significantly smaller. It also differs in terms of operating principle, in that the longitudinal velocity is not reduced uniformly across the width of the nozzle, neither is there a uniform increase in static pressure.

The flared nozzle relies, implicitly, on the swirl (tangential velocity) content of the primary flow, to effect a radial re-distribution of the axial flow momentum. In addition, the invention exploits the innate ability of swirling flows to undergo rapid widthways expansion (and contraction) without significant loss of swirl momentum, provided that the rate of expansion does not exceed a certain threshold governed by the flared nozzle geometry.

In physical terms, the outer part of the fluid flow is obliged to follow the convex curved profile of the flared nozzle due to the Coanda effect (see, for instance: *Massey, B. Mechanics of Fluids. 7th Edition, 1998, Stanley Thornes (Publishers) Ltd.*). Such a curved flow trajectory induces low pressure within the near-wall part of the flow, which causes the flow to adhere to the surface profile of the nozzle. The resulting progressive increase in outward radial flow within the nozzle is matched by the development of a radial pressure gradient within the body of the flow (reduction in axial pressure and a corresponding increase in radial [i.e. centrifugal] pressure), to maintain cyclostrophic balance. This, in turn, results in a corresponding reduction in longitudinal velocity and an increase in swirl (i.e. an increase in Swirl Number) within the body of the flow. The reduction in longitudinal velocity results in a corresponding increase in axial static pressure (i.e. an adverse pressure gradient).

It will be shown later that a similar effect can occur when a ring vortex encircles the flow. The core of the ring vortex forms a low-pressure centre, which induces an outward curvature in the longitudinal flow.

While the flared nozzle will achieve efficient rapid expansion with all types of swirling flows, the size of nozzle required to produce a highly expanded jet (needed for certain applications) can be greatly reduced if the inlet swirling flow is annular and has a relatively high swirl velocity. Accordingly, the second component of this invention consists of a swirl generator, designed to create an annular swirling flow with a Swirl Number of about 1. The swirl generator takes various forms depending on the particular application.

Because the present two-component invention relates to the production of swirling fluid flows and jets, particularly jets, having flow characteristics intended for specific tasks, it is important to be able to characterise the jet in terms of its (expanded) velocity components. Jet Swirl Number provides such a measure, and is defined in Appendix A. Jet Swirl Number can also be related to the flow topology (internal flow structure) and jet behaviour, which are discussed shortly.

In one embodiment of the present invention the flared nozzle and annular swirl generator are used in combination to produce a fluid flow or jet having a Jet Swirl Number of at least about 2, preferably about 4. Such flows and jets are characterised by a strong reverse circulation and are useful in combustion apparatus, in seabed excavation apparatus and in cleaning apparatus. In a second embodiment of the present invention the flared nozzle, without the annular swirl generator, is utilised to produce a fluid flow or jet having a Jet Swirl Number varying from 0.5 to 1.0, preferably from 0.7 to 0.9. Such flows are useful in enhanced-thrust, ducted propulsion, apparatus. In this case, the primary swirling flow is generated by a modified ship's propeller. In a third embodiment of the present invention, the annular swirl generator is used, without the flared nozzle, to produce a jet with a Swirl Number of about 1. Such flows are useful also in seabed excavation apparatus and in cleaning apparatus. Other embodiments of the flared nozzle principle can be used in existing equipment, which utilise swirling flows, to moderate or manipulate the outlet and/or internal flow and to improve operating efficiency.

It will be shown that by utilising the two components (flared nozzle and annular swirl generator) together, or separately, not only is considerable flexibility in operation possible, but that a number of novel applications can be realised.

The present invention will now be described in further detail, by way of example only, with reference to the accompanying diagrams in which:

Figure 1 is a perspective view of an embodiment of a nozzle in accordance with the first aspect of the present invention;

Figure 2 is a perspective view of a first embodiment of a swirl generator in accordance with the second aspect of the present invention;

Figure 3 is a perspective view of a second embodiment of a swirl generator in accordance with the second aspect of the present invention;

Figure 4 is a plot illustrating velocity profile associated with an annular swirling flow;

Figure 5 is a plot illustrating velocity profile associated with a more typical consolidated swirling flow;

Figure 6 is a plan view of the impeller of the embodiment of Figure 3;

Figure 7 is a cross section along line VII – VII of Figure 6;

Figure 8 is a perspective view of a second embodiment of a nozzle in accordance with the first aspect of the present invention deployed in a ducted propeller assembly;

Figures 9 and 10 show sectional and rear view elevations, respectively, of the ducted propeller assembly.

Figures 11 and 12 illustrate variations in jet form obtained with the embodiment of Figure 8 at slow and fast forward vessel speeds respectively;

Figure 13 is a perspective view of an embodiment of an excavating device incorporating the embodiments of Figures 1 and 3;

Figure 14 is a perspective part detail of Figure 13;

Figure 15 is a perspective view of an embodiment of a cleaning device incorporating the embodiments of Figures 1 and 2;

Figure 16 is a part cross-section of the embodiment of Figure 15;

Figures 17 and 18 show a fully broken high swirl number re-circulating jet, as would be produced by the Figures 13 and 15 respectively;

Figure 19 shows a simple modification to the Figure 2 embodiment to enable it to be used for combustion or for spraying purposes;

Figure 20 shows a detail of the near-nozzle portion of the jet produced by the Figure 19 embodiment;

Figures 21,22 and 23 are cross-sectional views illustrating how the flared nozzle and static swirl generator concepts can be incorporated into a Ranque-Hilsch vortex tube;

Figures 24 to 26 show plots of an elliptical curve obtained according to the stepwise procedures described herein; and

Figure 27 is a plot comparing area expansion ratio and inlet swirl number.

Figures 24 to 27 are also included as an appendix to illustrate particular design features of the flared nozzle (component A) and its geometry.

Starting with Figures 1, 2 and 3 there are shown the basic features of this two-component invention. Component A comprises the flared nozzle, which consists of a single part that is generally tubular in form with an inlet 1, a profiled body 2 and an outlet 3. Note that in Figure 1 only the inner surface profile of the flared nozzle is shown in order to indicate the general shape of the flare. Note that while the preferred profile is smoothly rounded (as indicated in Figure 1) for certain large-size applications a multi-facetted profile (made up from a series of conic rings) may be perfectly acceptable. The criterion for acceptability is that the flow should conform to the profile without significant separation from the surface. In practice, the outer surface may be parallel to the inner surface or it may be more uniformly cylindrical.

When in use, the flared nozzle would be attached, co-axially, by means of inlet 1 to a duct 4 emitting a swirling flow. Note that any type of swirling flow may be used, with the flared nozzle being designed accordingly, provided that the Swirl Number of the flow is known.

For certain applications of this invention, a particular type of swirling flow is preferred, which is provided by an annular swirl generator. This is component B of the invention. The annular swirl generator comprises a duct 4 to which the flared nozzle (component A) may be attached. Note that a variety of attachment means are possible, including direct attachment, flange attachment, screw attachment, etc.

Figures 2 and 3 show two alternative types of annular swirl generator, which are referred to herein as components B1 and B2. Component B1 is described as a passive annular swirl generator, in that it relies on an external source of pressured fluid (shown arrowed). The pressurised fluid is directed, via a suitably enclosing sleeve 5, through holes 6 drilled on planes orthogonal to duct 4 and disposed tangentially, so that the flow is forced to travel around the circumference of recessed inner (swirl) chamber 7. Note that holes 6 may be circular (as in Figures 2, 16 and 19), or in the form of narrow longitudinal slits of equivalent open area (as in Figures 21 and 22). Tangential holes 6 are preferably positioned so that the flow entering inner chamber 7 does so without mutual interference. This is achieved by drilling the four holes as shown in Figure 2 in pairs at two levels, such that the holes are positioned at 90° relative to one another. Preferably also, an even number of holes is drilled to achieve a balance of flow through the duct. Further details of the form of construction of component B1 are given in WO2004/045775. The highly-swirled flow generated in swirl chamber 7 then spills out into duct 4 in the form of an annular swirling flow, with a velocity profile similar to that shown in Figure 4. Note that the longitudinal component (v_{long}) of the flow essentially results from the passage of fluid through the duct. The Swirl Number of the flow is thus determined by the ratio of the aggregate surface area of the tangential holes, to that of the duct. This alternative, Geometric Swirl Number, is also defined in the Appendix.

Because of the nature of the swirl generating process, the longitudinal velocity profile produced by component B1 (and component B2) is not uniform, rather the velocity is higher towards the periphery, as is shown in Figure 4. The increase in swirl (v_{swirl}) and longitudinal (v_{long}) velocities away from the axis of the duct and the near coincidence of their maxima towards the periphery of the duct, is a characteristic of the annular swirling flow produced by this invention. (see for instance: *Shtern, V., Hussain, F. and Herrada, M. New Features of Swirling Jets. Physics of Fluids, 2000, 12, pp 2868-2877*).

Alternative annular swirl generator B2 is referred to as an active swirl generator because it incorporates a flow-driving mechanism in the form of motor 8 and rotating impeller 9. Note that motor 8 may be powered electrically, hydraulically or pneumatically. A compact motor form is preferred, with a direct drive to the impeller so that the motor can be housed co-axially within duct 4 or, as shown in Figure 3, within the expanded bellmouth intake 10 to the duct, by means of support fins 11.

Note that the form of component B2 as shown in Figure 1 is essentially the same as the ducted propeller described in WO2004/045775 and PCT/GB2004/000309. However, instead of using a conventional Kaplan propeller (with or without a disc to reduce the axial component of longitudinal flow) the present invention uses a specific design of impeller 9. The term impeller is used (as opposed to propeller) to indicate that the present invention is not, primarily, a means for generating axial flow and propulsive thrust. Rather, impeller 9 is designed to produce a swirling flow: specifically, an annular swirling flow, with a velocity profile as shown in Figure 4. Because it induces a significant radial flow component, impeller 9 may be seen as being similar to the impeller of a mixed-flow pump.

A propeller (such as conventional ship propeller or the Kaplan propeller described in WO2004/045775) produces a flow with a velocity profile similar to that shown in Figure 5. Such a profile (adapted from: *Hughes, M.J., Kinnas, S.A. and Kerwin, J.E. Experimental Validation of a Ducted Propeller Analysis Method. Journal of Fluid Engineering, Transactions of the ASME, Vol. 114, June 1992, pp 214-219*), where the axial and swirl velocity maxima are spatially separated and there is a concentration of swirl velocity towards the axis, is less readily expanded by means of the flared nozzle. Note that in Figure 5, an increase in swirl velocity towards the axis is characteristic of a free-vortex type of swirling flow produced by conventional marine propellers. Close to the flow axis, the local increase in longitudinal velocity and reduction in swirl velocity are characteristic of a forced-vortex type of swirling flow. With conventional marine propellers this near-axis zone corresponds to the hub vortex (or vortex core jet, see Appendix).

Conventional propulsive propellers are normally designed to suppress both the swirl and radial velocity components, because the latter do not contribute directly to thrust. Consequently, conventional propulsive propellers are not ideal for creating a readily expandable swirling flow for those applications where a high Jet Swirl Number is required. Conversely, the inherent reluctance of such relatively low-swirl flows to expand can be advantageous where a long columnar jet is required (e.g. for stand-off jetting, see WO2004/045775).

Impeller 9 is shown in more detail in Figures 6 and 7. It is fabricated out of a circular sheet of metal (stainless steel being an appropriate material for use underwater), as is shown in template form in Figure 6. A taper 12, better seen in Figure 5, is machined on the outer part of the upper surface destined, in part, to become the blades. Note that the thickness of the plate and the angle of the taper (4.5°) is designed (in similar fashion to a conventional propeller) to provide adequate mechanical stiffness, while limiting the flow obstruction created by the leading edges of the blades. Full depth 13, and partial depth 14, slots are cut in the upper surface of the plate. These slots define both: the pointed arms 15 of an eight-cornered radial star pattern, and the outline of the blades 16. Note that an eight-cornered star provides a convenient compromise between blade area and blade root sectional area. It is conceivable that a seven-corner star might also be used. Circle 17, which connects the inner points of the star and also corresponds with the inner edge of the taper, has a diameter approximately 40% of the finished diameter. The area of this inner circle, together

with the aggregate area of the arms of the eight-pointed star 15, is equivalent to a circle approximately 60% of the finished diameter of the impeller. Note this is the same as the size of the circular disc described in WO2004/045775, as being the optimum for increasing the swirl number of a conventional Kaplan propeller. The reason for this size of disc will become apparent shortly.

The additional parts of the plate that need to be trimmed away to form the impeller are indicated in Figure 6. Note that this trimming operation produces a stepped outer diameter 18, while a triangular portion 19 of each blade trailing edge is removed. The profile of the blades is then formed by bending, along the partial depth slots 14, so that the blades adopt an angle of intersection of approximately 144° to the underside of the plate (36° downwards from the horizontal). The V-shaped grooves that open up along slots 14 during this bending procedure are then in-filled with weld metal, to give a rounded profile. Note that as the blades are bent down, the outer diameter 18, as trimmed, converges to the circular diameter of the finished impeller. A slight excess of metal is allowed on the outer diameter, so that the impeller can be machined to an exact coaxial fit inside the duct, leaving a minimum of gap clearance. A sectional view of the finished impeller (only two blades being shown) is shown in Figure 7. Figure 7 also shows the hub mounting arrangement 22, which enables the impeller to be attached to the tapered motor shaft. Note that a taper and keyway combination is the preferred (but not sole) method of attaching the impeller to the motor shaft.

There are three features of impeller 9, which make it particularly suitable for generating the annular swirling flow profile shown in Figure 4:

1. The central part of the impeller (defined by circle 17) completely blocks the axial longitudinal flow, while the star-shaped arms 15 that support the blades 16 have an outwardly diminishing blocking effect. Longitudinal flow is thus only permitted through the inter-blade openings in the outer part of the impeller disc.
2. The rotating non-blade parts of the disc induce swirl (solid body rotation) into the flow (more so than with a conventional propeller). They also cause the blocked longitudinal flow to be deflected radially outwards as a result of the no-slip boundary condition on a rotating disc (see for instance: *Escudier, M.P. Observations of the flow produced in a cylindrical container by a rotating endwall. Exp. Fluids, 1984, 2, pp 189-*). Longitudinal flow drawn from upstream towards the impeller disc thus diverges towards the outer parts of duct 4 before it passes through the disc. Note that the axial part of the underside of the impeller disc also acts like a rotating endwall, imparting swirl and a centrifugal force to the water in downstream contact with the disc.
3. Because the blades 16 are bent along a tangent line 14, rather than a radial line, they impart a component of radial-outward directionality to the flow as it passes through the impeller disc. This further serves to concentrate the longitudinal and swirl velocity components towards the outer parts of duct 4. The higher longitudinal velocity means that the blade pitch (angle of the blades) is, accordingly, greater than in a conventional propeller. Nevertheless, because of the centrifugal (outward radial) forcing, the outer parts of the blades also operate in a

higher pressure-field, so there is no greater cavitation risk than with a conventional propeller.

Note impeller 9, in the form described, is designed to create a flow with a Swirl Number of 1. This is significantly higher than with a conventional propeller (where the Swirl Number is, typically, about 0.3) and with the high-swirl Kaplan propeller (operating at zero advance) described in WO2004/045775, where the Swirl Number is about 0.4-0.5. The advantage of the higher Swirl Number is discussed later.

Component A, the flared nozzle, is designed to operate with fully turbulent inlet flows having Swirl Numbers typically in the range 0.2 to 2.0. Many practical flows have Swirl Numbers in this range, or swirl generators of the type described may be used to create such a swirling flow. Provided the flow is turbulent, the flared nozzle is relatively insensitive to the Reynolds number of the inlet flow (see Appendix for definition). The nozzle can achieve a maximum expansion, in terms of inlet-to-outlet area ratio, of 4 times, which is equivalent to a 2 times diametral expansion. Thus to produce a jet with a Jet Swirl Number of 4 the inlet Swirl Number has to be 1 or higher. It is for this reason that components B1 and B2, which are designed to produce a flow with a Swirl Number of 1, constitute an integral part of this invention.

To achieve optimum (i.e. maximum rate) expansion of a swirling flow through such a flared nozzle, the longitudinal profile of the inner surface of the nozzle has to be elliptical (parabolic) in form. Figure 1 shows such a profile, while Figure 24 illustrates a means for simple geometrical construction of an elliptical profile. An elliptical profile is fundamental to the design and operation of the present two-component invention. The dimensional characteristics of the elliptical flared nozzle are determined, step-wise, according to a set of rules, which are set out as follows.

Step 1

The Area Expansion Ratio of the nozzle: that is ratio of outlet-to-inlet areas, is first determined. This defines the change (i.e. increase) in Swirl Number through the nozzle.

Area Expansion Ratio (AER), is defined as:

$$AER = \frac{a_2}{a_1} \qquad \text{(Rule 1.)}$$

Where: a_1 = cross-sectional area of inlet
a_2 = cross-sectional area of outlet

Change in Swirl Number through the nozzle is equal to the AER of the nozzle. Note that AER can be up to 4.

Step 2

Area Expansion Ratio then allows the step height of the nozzle to be determined. Step height (H) is defined as:

$$H = (d_2 - d_1) \div 2 \qquad \text{(Rule 2.)}$$

Where: d_1 = diameter of inlet
d_2 = diameter of outlet

Step 3

Nozzle length is then determined. Where nozzle length (L) is defined as:

$$L = 2H \div \text{inlet Swirl Number} \qquad \text{(Rule 3.)}$$

Step 4

The appropriate elliptical curve can then be drawn using the previously determined step height and nozzle length dimensions as orthogonal axes. Any elliptical curve-drawing method may be used; the one shown in Figure 24 uses a ruler and length of string or thread and adopts the following equations:-

Separation distance $= \sqrt{(L^2 - H^2)} \quad x\,2$

String length $= 2 \times L$

In Figures 25 and 26 a number of elliptical profiles are plotted, to illustrate some key features. In the case of Figure 25, the nozzle diameter, step height and AER are kept fixed, but the inlet Swirl Number is varied. In the case of Figure 26, the nozzle diameter and inlet Swirl Number are kept fixed, but the AER is varied. Referring to Figure 25: note that as the inlet Swirl Number decreases, the nozzle length (for optimum expansion) increases, rapidly so at low inlet Swirl Numbers. Note also, that at an inlet Swirl Number of 2 the nozzle becomes uniquely circular in profile, while at still higher inlet Swirl Numbers the nozzle profile becomes elliptical again, but with the step height being greater than the length (axes reversed). With fixed nozzle diameter and inlet Swirl Number (Figure 26) both nozzle length and nozzle outlet diameter increase with increasing AER.

The rapid increase in nozzle length at low inlet Swirl Numbers, for a fixed step height, is because nozzle length is inversely proportional to inlet Swirl Number (parabolic function). The product of step height and nozzle length, on the other hand, is directly proportional to AER (linear function). Note that at very low Swirl Numbers the nozzle length becomes very large, and comparable to that of conventional conical diffusers. Since a Swirl Number of 0.05 can be taken as a practical lower limit for self-sustaining swirl motion in pipes (below this swirl damps out very quickly), a step height to nozzle length ratio of 1:40 can be seen as the point where the flared nozzle and conventional conical diffusers overlap.

Area Expansion Ratio and inlet Swirl Number, together, define unique flared nozzle geometrical characteristics. This is shown in Figure 27, where optimum AER and inlet Swirl Number are plotted in AER/nozzle-geometry space, non-dimensionalised by inlet Swirl Number. Figure 27 is, in effect, a graphical form of Rules 1.) to 3.), and can be used to derive nozzle geometry for the range of AER and Swirl Number values plotted. Figure 27 illustrates another key feature, which is that for each inlet Swirl Number a flow can be expanded in optimum fashion to a maximum of AER = 4, since the latter marks an asymptotic boundary on the graph.

Published experiments on expanded swirling flows in closed pipes (see for instance: *Dellenback P. A., Metzger D.E. and Neitzel G.P. Measurements in Turbulent Swirling Flow Through an Abrupt Axisymmetric Expansion. AIAA Journal Vol. 26. No. 6, June 1988*, and *Guo B., Langrish A.G. and Fletcher D.F. Simulation of Tubulent Swirl Flow in an Axisymmetric Sudden Expansion. AIAA Journal Vol. 39, No. 1, January 2001*) provide a basis for understanding how the flared nozzle works, and the effects of different Swirl Numbers, as they manifest themselves in a swirling fluid flow or jet. Previous publications do not, however, recognise the significance of an elliptical profile in providing a natural (optimum) bounding surface to the expanding flow.

To appreciate the potential uses of this device it is necessary to have a basic knowledge of the flow characteristics associated with fully turbulent swirling flows and jets. Table 1 (incorporating published data from *Dellenback et al 1988* and *Guo et al 2001*) shows how an expanded swirling jet will behave based on its Jet Swirl Number. Jet Swirl Number has been derived from the above referenced data, by multiplying the inlet Swirl Number by the appropriate Area Expansion Ratio. Note that Jet Swirl Number is a convenient way of quantifying swirling jets for the purpose of rationalising their behaviour. It represents the Swirl Number of the expanded flow as it emerges into the ambient (or into a larger diameter duct). For completeness, the behaviour of a non-swirling jet is also included.

Table 1.

Jet Swirl Number	**Jet Behaviour**
0	Jet is axi-symmetric. Reynolds boundary stresses are strongly negative so jet entrains ambient fluid as it expands width-ways, at an included angle of 20°-25°.
0-0.5	Jet becomes increasingly columnar (straight-sided), Reynolds boundary stresses reduce and entrainment diminishes*. Precession (spiral rotation) about the jet axis takes place, although the axis of precession departs only slightly from the main jet axis.
0.5-1.5	Precession starts to become increasingly irregular, with a higher order precession superimposed on the main precession. Speed of rotation of the higher order precession increases linearly with increase in Swirl

	Number. Axis of main precession departs increasingly from the jet axis to produce a straight-sided conical jet. Maximum cone angle is about 60°, corresponding to the highest Swirl Number. Reynolds stresses become negative on the inner surface, so entrainment takes place there; but positive on the outer surface where detrainment takes place. Ambient flow towards the inner surface occurs uniformly across the width of the jet and this flow is virtually swirl free. By implication, the apex of the cone is marked by a steady out-of-balance inducing disturbance, which may be located axially (lop-sided spherical re-circulation "bubble") or peripherally (partial or lop-sided ring vortex), or a combination of both. The disturbance(s) grow(s) with increase in cone angle. Jet also experiences increasing pressure fluctuations, which peak at a Swirl Number of 1.
1.5-2.0	Jet experiences full breakdown, with transformation into a two-cell axi-symmetric jet, initially with the same 60°-cone angle. Axial steady disturbance (spherical breakdown "bubble") becomes symmetrical and with a diameter greater than the jet vortex core, or equal to the jet vortex core if a peripheral steady disturbance (symmetrical ring vortex) is present.
2.0-4.0	Jet form remains essentially as above, but re-circulation "bubble" becomes elongate and develops an extended axial counter-flow tail. Axial counter-flow tail develops into an annular jet flow with increasing Swirl Number, and is accompanied by a strong centrifuging effect at the head of the counter-flow stream. Between the outer conical jet and the axial counter-flow annular jet, the flow is essentially re-circulatory. Re-circulation is driven by the outer conical jet flow along a pronounced shear surface, which separates the two flow regions. This shear surface appears to develop secondary oscillations (eddies). With increasing jet swirl number the angle of divergence of the conical jet increases.

*Note, however, if there is a significant increase in vortex strength towards the axis of the jet (e.g. strong hub vortex, as shown in Figure 3) this can act as a sink, causing entrainment from the outer part of the jet and overall contraction of the jet.

Note also that the steady axial and peripheral disturbances, which generally accompany the varying jet topologies (as described in Table 1), are those that would tend to occur, more particularly, with confined flows. In confined (i.e. sudden expansion pipe) flows, unconstrained outward spreading of the jet downstream cannot occur, and is replaced by smearing of the flow against the pipe wall. In sudden pipe expansion flows the steady peripheral disturbance generally takes the form of a corner eddy (equivalent to a ring vortex).

While the changes in jet form described in Table 1 essentially form part of a continuum, a Swirl Number of 1.5 (or to be exact, $1.414 \equiv \sqrt{2}$) can be seen as marking a critical threshold in terms of internal flow dynamics. (See for instance: *Benjamin, T B. Theory of Vortex Breakdown Phenomena. J. Fluid Mech. Vol. 14, 1962, pp 593-629* and *Billant, P., Chomaz, J-M, and Huerre, P. Experimental study of vortex*

breakdown in swirling jets, J. Fluid Mech. 1998, Vol. 376, pp 183-219). Up to this point, the primary longitudinal jet flow remains essentially intact, albeit departing significantly from the jet axis in the form of a spiral precessing conical jet.

The form of the latter jet is shown in Figures 11 and 12 (see also: *Novak, F., and Sarpakay. T. Turnbulent Vortex Breakdown at High Reynolds Numbers. AIAA Journal, 2000, Vol 38, No. 5, pp 825-834*, for a description of this type of jet). At the critical threshold, however, the swirl effectively overcomes the axial flow and the jet undergoes full breakdown, changing from an essentially asymmetric single-cell flow to an axi-symmetric two-cell flow, with the development of a pronounced axial counter-flow and re-circulation region within the secondary flow field. This transformation is analogous to a hydraulic jump (in surface flows) or normal shock (in compressible flows). With increasing Swirl Number above the critical value, the jet is able to cone out at a wider angle, and a progressively stronger counter-flow develops along the axis. Such a re-circulating jet is shown in Figures 17 and 20.

Note that in many of the published experiments on vortex jet breakdown in confined flows, it is only the narrow central vortex core that experiences this type of breakdown, the outer part of the swirling flow typically remains unaffected. Note also that the narrow central vortex core is essentially an axial forced-vortex jet (see Appendix A for definition of forced vortex). Such a narrow vortex jet core zone is indicated in Figure 5 for a propeller jet. In this case, it is often referred to as the hub vortex because it appears to emanate from the downstream end of the propeller hub.

The steady axial disturbance (referred to in Table 1) can be seen as applying a pressure against this core jet flow (see, for instance: *Marshall, J. S. The Effect of Axial Pressure Gradient on Axisymmetric and Helical Vortex Waves. Phys. Fluids, A 5, (3), March 1993, pp 588-599* and *Darmofal, D.L., Khan, R., Greitzer, E.M. and Tan, C.S. Vortex Core Behaviour in Confined and Unconfined Geometries: a Quasi-one-Dimensional Model. J. Fluid Mech., Vol. 449, 2001, pp 61-84*). The effect is analogous to a ball being pressed against the outlet stream from a pipe. The jet core is deflected to form a conically splayed jet. Up to the point of full breakdown, the steady axial disturbance (the ball) has been smaller than the jet core (the pipe) and has tended to oscillate, causing the jet to cone out but in spiral fashion (c.f. a rotating garden sprinkler). With increasing Swirl Number beyond the breaking point, the creation of an axial counter-flow stream reinforces the axial blocking effect and leads to the development of a secondary re-circulation flow pattern. Note that the role of the steady axial disturbance can be augmented by that of a steady peripheral disturbance, which in confined vortex jet experiments is achieved by means of an expansion of the pipe. This expansion of the flow has the effect of reducing the axial velocity and thereby creating an adverse pressure gradient (i.e. an increase in pressure) along the axis. Note this increase in axial pressure is relative to the already low pressure within the flow created by the swirl.

In the case of the present invention, the core zone of the duct flow may vary considerably in width in relation to the duct diameter, depending on the mode of operation and the Swirl Number of the primary flow. For instance, it may comprise a

relatively narrow axial zone, as shown in Figure 5 in the case of the low swirl flow from a conventional propeller. Or it may extend over nearly the full width of the duct, as shown in Figure 4, when the B1 and B2 component high-swirl generators are used. Given that to achieve the duct flow profile shown in Figure 4, with a Swirl Number of 1, already requires a (near) 60% blocking of the width of the duct (see earlier reference to fore-propeller disc), any further increase in Swirl Number by this blocking mechanism would not be feasible. Therefore, the steady peripheral disturbance (in this case emulated by component A: the flared nozzle), is used to overcome the need for the steady axial disturbance (the blocking effect) to increase in size. The flared nozzle and annular swirl generators can thus be seen as:

1. Supplanting the natural steady disturbances (corner eddy and axial re-circulation bubble, respectively);
2. Having overlapping roles, and;
3. Working in concert, to enable a very wide spectrum of Jet Swirl Numbers and jet flow characteristics to be produced.

By means of the flared nozzle, any of the flow characteristics described in Table 1 can be induced, simply by selecting a suitable swirl generator that produces a flow with a known Swirl Number, and by matching the flared nozzle geometry according to the aforementioned rules. However, in order to keep the flared nozzle to a practicable size and yet be able to produce Jet Swirl Numbers up to 4, a means for generating an initially quite high Swirl Number flow is required. It is for this reason that the invention includes the component B1 and B2 swirl generators, for generating an annular swirling flow with a Swirl Number of 1. It should be appreciated that the induced flow patterns develop inside the flared nozzle and that the change of flow pattern occurs spontaneously as the Swirl Number of the flow changes.

A number of potential applications for the two-component invention will now be briefly described, covering a range of inlet and Jet Swirl Numbers. The examples also serve to illustrate both application of the flared nozzle design rules and utilisation of the jet characteristics outlined in Table 1. The examples given are listed in order of increasing Jet Swirl Number.

By expanding any lesser Swirl Number flow to give a Jet Swirl Number of greater than 0.5 but less than 1.0, a conical spiralling jet, with a cone angle up to about 45^0 can be produced. In such a conical jet, longitudinal flow momentum is coupled into the ambient fluid not only over the outer surface of the cone, but potentially over the inner surface also. Contrast this with to a columnar swirling jet or a non-swirling jet, where interaction with the ambient fluid can only take place over the outer part of the jet. With the conical jet (in contrast to the columnar jet) interaction also occurs over a progressively larger surface area because of the downstream increase in diameter of the cone.

Rapid coupling of jet energy into the ambient fluid provides a basis for enhanced (jet) thrust in rotational-based (i.e. propeller) propulsion systems and is particularly appropriate for slow-speed vessel operation. However, the cone angle has to be kept

relatively small because the flow towards the inner cone surface is essentially a counter-flow, and can induce increasing negative thrust at large cone angles. Also as vessel speed increases, relative to jet flow speed, this counter-flow and the cone itself create an increasing amount of drag.

It is worth noting that in the far wake of a normal ship's propeller a similar, but less pronounced, spiral coning effect often occurs (see, for instance: *Stella, A., Guj, G., Di Felice F. and Elefante, M. Experimental Investigation of Propeller Wake Evolution by Means of Flow Visualisation. Journal of Ship Research, Vol. 44, No. 3, September 2000, pp 155-169*). This is generally described as the "wake ageing" effect, although its cause can be attributed to an increase in Swirl Number as the longitudinal flow decelerates and the jet changes from a consolidated jet (Figure 5) to a more annular jet (Figure 4). The flared nozzle, by inducing a more pronounced conical spiral jet, can be seen as providing a means for significantly foreshortening the slipstream ageing process. By both increasing the rate of longitudinal (axial) flow deceleration and attenuating the overall length of the slipstream, the flared nozzle also enables vessels to operate more effectively in confined waterways, with better manoeuvring and less propeller-wash damage to harbour structures.

Conventional ships' propellers normally operate in the Jet Swirl Number range 0.25-0.5, depending on the speed of advance. The highest Jet Swirl Number occurs at zero advance speed. Table 1 indicates that at a Swirl Number of 0.3 the jet is almost columnar with little tendency to interact with the ambient fluid. Transfer of momentum between jet and ambient fluid, which is a necessary part of thrust development, thus takes place over a long jet distance. Even though this distance is foreshortened when the vessel is operating at slower speed, due to the increase in Swirl Number associated with less flow through the propeller, nevertheless the jet still shows little tendency to spread. For slow-speed operation, therefore, a long columnar jet can be seen as rather inefficient, particularly if the propeller is of small diameter (less jet surface area). In practice, the jet may also tend to wander, resulting in a loss of axial thrust directionality; also boundary effects (e.g. dock structures) and changes in vessel direction (causing the jet to become curved) often conspire to further reduce the efficiency of energy coupling and forward thrust.

By adding a flared nozzle to a conventional ducted-propeller assembly and raising the Jet Swirl Number above 0.5, slow-speed thrust efficiency can be enhanced. However, Figure 23 shows that for an inlet swirl number of less than 0.5 the flared nozzle will be quite long (nearly four times the step height) to achieve any significant expansion. A modified propeller design is thus required to keep the duct and flared nozzle to a practicable size. The form of this modified propeller and duct/flared nozzle arrangement is shown, by way of example, in Figures 8,9 and 10.

Propeller 23 appears, outwardly, similar in shape to a conventional 4-bladed, large blade area propeller, normally used in ducted propulsion systems. However, while conventional propellers typically have the pitch line 24 (see Figure 10) corresponding to a radial line, in this case the pitch line can be rotated forward angle to about 26 from the radial to position 25 in Figure 10. This creates what is termed "unbalanced

forward skew", as indicated by the dashed outline of the blades in Figure 10. The net effect is that the blades impart an increasing outward radial velocity component to the longitudinal flow through the propeller disc as the advance angle is increased. Confined by the duct, this radial flow has the effect of concentrating both the longitudinal and swirl velocity maxima towards the periphery of the duct. Since radial velocity is produced at the expense of longitudinal velocity, there is also a resulting net increase in Swirl Number.

In order to be able to control the forward advance angle of the blades, each blade 26 is independently mounted on a pin 27 within hub assembly 22. Pin 27 is attached to shaft plate 28, which forms part of the drive shaft 29. In the embodiment shown in Figure 8 drive shaft 29 is enclosed by housing 30 and supported on bearings 31. Blade boss 32 through which pin 27 passes is further enclosed by rear plate 33, also secured by pin 27. Note that boss 32 can rotate on pin 27 but a minimum sliding gap clearance is allowed between adjoining faces 34 and 35 (in Figure 9).

An elastomeric seal 36, cemented to the sides of each blade boss and compressed between plates 28 and 33, provides enclosure to the inner part of the hub assembly. The elastomeric seal is designed to distort as the blade angle of advance chages, as indicated in Figure 10. To complete the sealing of the inner part of the hub assembly, O-ring seals 37 are provided, recessed into grooves in plates 28 and 33.

Drive shaft 29 is hollow and has rod 31 co-axially located through it. Rod 38 has worm 39 machined onto the end located within the hub assembly. It also has a keyway slots 40 machined in the distal end, which locate onto corresponding key projections within housing 41 that forms an extension to plate 33. Keyway assembly 40 causes rod 38 to rotate with drive shaft 29, but does not prevent rod 38 from sliding in and out of drive shaft 29. Controlled in and out movement of rod 38 (indicated by the double-headed arrow in Figure 9) is effected by a separate mechanism (not shown) at the proximal end of drive shaft 29.

In and out movement of rod 38 causes spur gear wheel 42 to rotate. This rotation occurs because spur gear wheel 42 has a corresponding thread machined on its inner bore and fore-aft movement is constrained by plates 28 and 33. A fixed amount of in-out movement of rod 38 causes a corresponding fixed amount of rotation of spur gear wheel 42. Spur gear teeth 43 on gear wheel 42 mate with similar teeth 44 on blade boss 32 and convey rotation of gear wheel 42 to blades 26, which rotate in unison.

Propeller 23 also has a slightly unconventional pitch profile. Instead of the pitch angle decreasing outwards along the blade span (as in conventional propeller design, where the blades can be seen as being twisted), approximately the same pitch angle (26°) is maintained from boss 32 to tip 45. When the blades are in the zero advance position (no radial flow generation), a more even swirl profile and an accentuated annular longitudinal velocity profile is produced. The velocity profile is more or less midway between the velocity profiles shown in Figures 4 and 5. When the blades are advanced, the velocity profile is biased more towards the profile shown in Figure 4.

Note the overall similarity of this effect to that created by impeller 9 in the component B2 swirl generator shown in Figure 3. Note also that to accommodate the hub assembly components, as described, the hub diameter is larger than with a conventional propeller, being approximately 40% of the propeller diameter. This enlarged hub and the conical expansion of housing 30 helps to discourage the formation of a hub vortex.

It is important to appreciate that this modified propeller is designed to work, specifically, in a duct. It will not work in a free-stream situation, particularly with the blades advanced, because the radial forcing causes the water flow through the propeller disc to be flung out sideways. The duct is, therefore, required to contain and re-direct this radial flow momentum in a more longitudinal direction. This modified propeller is thus rather different from conventional ducted propellers, which will also work tolerably well in a non-ducted situation.

Duct 4, in which the modified propeller is coaxially mounted, is either of the straight-sided (tunnel) type or (as shown in Figure 8) of the Kort nozzle type (see, for instance: *van Manen, J.D. and Superina A. The design of screw-propellers in nozzles. Int. Shipbuilding Progress, March 1959, Vol. 6, No. 55*). This latter type of duct also induces an additional component of longitudinal velocity, which is concentrated within the propeller tip region of the duct. This duct-induced flow is a further reason for the higher pitch of the outer part of the blades.

Note that the outer tip 45 of each blade 26 is designed to follow the profile of the duct as the advance angle of the blades changes.

The flared nozzle (component A) may be attached directly to the aft end of the duct (or Kort nozzle), if only slow speed operation is required. For a particular vessel propulsion application, the flared nozzle and modified propeller combination would be matched to the specific (slow) vessel design speed that it was intended to enhance. This is referred to as the Nozzle Design Speed. For such slow speed operation, the jet 46 (see Figure 11) would be designed to cone out at a specific angle (of up to 45°) in order to achieve rapid coupling of momentum into the ambient fluid over a short wake distance. By operating at a Jet Swirl Number of about 0.9 advantage can also be taken of the high pressure developed within the cone-shaped jet, as noted in Table 1.

In the case of forward propulsion at higher speed (with the blades set at a low or zero advance angle), the influence of the flared nozzle will be reduced (i.e. the jet cone angle will diminish), increasingly so as forward vessel speed increases. This is because of the additional component of longitudinal flow through the duct, causing both flow-separation from the inner surface of the flared nozzle and an overall reduction in Jet Swirl Number. In order to promote additional flow separation and attendant reduction in jet cone angle, the flared nozzle (Component A) may be attached to the duct (Kort nozzle) by radially disposed plates 47, creating a gap 48 between duct 4 and flared nozzle (component A). Note this is similar to the van der Giessen type of "Wing Nozzle". As forward vessel speed increases, progressively more (non-swirling) flow is drawn through this gap. This causes a reduction in the

adhesion of the flow to the inner surface of the flared nozzle and thus dramatically reduces the expansive effect of the flared nozzle. The less conically expanded, higher longitudinal velocity, jet resulting from the diminished influence of the flared nozzle, is more suitable for high-speed forward propulsion. Such a jet 49 is shown in Figure 12. Note the similarity of the latter jet to the slender wake described by *Shtern, V., Borissov, A. and Hussain, F. Vortex sinks and axial flow: Solution and applications. Phys. Fluid. October 1997, Vol. 9 (10), pp 2941-2959*. With such a jet, momentum transfer takes place over a somewhat longer wake length (than the jet shown in Figure 9).

The propeller/duct/flared nozzle combination shown in Figure 8 works, in effect, rather like a road vehicle gearbox; with the gearing ratio set, in this case, to avoid the Swirl Number range over which the jet is most columnar and experiences the least interaction with the ambient fluid. The above described method of operation is seen as being particularly appropriate for vessels where variable forward motion is the norm, such as tugs and ferries; as well as for vessels and craft using azimuth thrusters and outboard motors.

The above applications relate to jets, which are not fully broken. Other applications of such conical spiralling jets are described in WO2004/045775.

Once fully broken, jet behaviour changes completely and the jet becomes potentially useful for a different set of applications, depending on its breakdown state. For the applications now to be described, a Jet Swirl Number greater than 2 would typically be used.

With increasing Jet Swirl Number above 2, a progressively greater amount of axial counter-flow and re-circulation takes place within the body of the jet, the jet also splays out at a progressively wider angle due to centrifuging at the head of the counter-flow stream. Additionally, there is more internal shear, and so greater turbulence within the jet. Applications, which currently make use of these characteristics, include: swirl combustors as used in gas turbine engines (see, for instance: *Shin, H-W. Laser Doppler Anemometer Measurements in Re-Circulating Flowfields. In: D.E. Ashpis et al (Eds). Instabilities and Turbulence in Engineering Flows, Kluwer Academic Publishers, 1993, pp 293-306*), and gas- and oil-fired boilers; spray dryers (see, for instance: *Guo B., Langrish A.G. and Fletcher D.F. Simulation of Tubulent Swirl Flow in an Axisymmetric Sudden Expansion. AIAA Journal Vol. 39, No. 1, January 2001*), the Ranque-Hilsch vortex tube and a novel seabed excavation device, as described in WO2004/045775 and PCT/GB2004/000309.

The novel seabed excavation device, whose development led to the realisation of the elliptical flared nozzle concept, will now be described in more detail. This particular device utilises the strong axial counter-flow stream and well-developed re-circulation flow pattern generated by a highly swirled broken jet (Jet Swirl Number of between 2 and 4), to both erode and lift bed material into suspension. Material transported upwards in the axial counter-flow stream is flung out sideways (centrifuge effect) into

the radially deflected annular flow; being transported away as this flow first descends to and then spreads out across the bed.

Figures 13 and 14 show, by way of example, one way in which components A and B2 can be used together for underwater excavation or vessel cleaning. Note that other embodiments of this combination are described in WO2004/045775. Component A, the flared nozzle, is designed for easy attachment and detachment by means of a bolted flange. This enables different sizes of flared nozzle (giving different Jet Swirl Numbers) to be used, or component B2 to be operated on its own. There are a number of reasons why component B2 may be operated on its own. For instance, in the embodiment shown in Figure 13, where it is attached to the arm 50 of a barge- or pontoon-mounted long-reach excavator, component B2 (without the flared nozzle) can provide a means for propelling and/or manoeuvring the barge or pontoon, rather like an azimuth thruster. Also, component B2 operating on its own with the aforementioned impeller produces a conical jet, which has particular uses for bed excavation (as described in WO2004/045775).

Figure 15 and 16 show, also by way of example, one way in which components A and B1 can be used in intimate combination for underwater hand-held cleaning. Note that this and other embodiments of this combination of components are described in WO2004/0457750.

The form of the free jet, created by the devices shown in Figures 13/14 and 13/16, is illustrated in Figure 17. How the jet form changes during impingement against the bed is shown in Figure 18. Note that salient features of the jet are also described in WO2004/0457750. Excavation, associated with jet impingement, reaches a maximum with a jet Swirl Number of 4. This is because:

1. The jet splays out at the widest angle, giving the maximum impingement footprint area.
2. The counter-flow 51 is strongest, which means that a strong suction is applied to the bed, causing fluidisation, and the maximum amount of material can be lifted upwards in the axial counter-flow stream.
3. Because of the increased density of the upward counter-flow stream (due to the concentration of suspended sediment) strong centrifuge action occurs at the head 52 of this flow stream. Sediment particles (together with an added mass of water) are flung out sideways into the radially deflected longitudinal flow 53 to be carried away by this flow. This causes a reduction in pressure at the head of the counter-flow stream, which in turn helps to drive the upward counter-flow and flow re-circulation 54. Centrifuging action also causes the whole jet to splay out more widely (jet becomes umbrella-shaped), with further increase in the size of the impingement footprint. A similar centrifuging effect (although in a confined fashion) is seen in rotating pipe heat exchangers (see, for instance: *Shtern, V., Zimin, V. and Hussain, F. Analysis of Centrifugal Convection in Rotating Pipes. Physics of Fluids, Vol. 13, No. 8, August 2001, pp 2296-2308*).
4. In the presence of frictional (i.e. non-cohesive) bed materials, a strong centripetal flow 55 (Eckman layer flow) takes place across the bed within the impingement

footprint region. Because this near-bed flow has increased turbulence, as well as viscosity (due to fine sediment in suspension), it is able to actively erode the bed and transport a wide range of sediment particle-sizes inwards, in suspension, towards the axial counter-flow zone. Note that the suction effect on the bed is reduced in the presence of this near-surface centripetal flow.

5. Because the material lofted in the counter-flow is entrained directly into the high velocity deflected longitudinal flow stream, very little material escapes from the umbrella-shaped envelope of the jet. The process, therefore, does not cause a significant increase in the amount of suspended sediment in the water column.
6. Because the peripheral jet flow stream meets the bed at a shallow angle the flow 56 continues across the bed without significant loss of momentum, and the high concentration of sediment particles means that this near-bed flow will propagate as a density current over long distances.
7. In addition, at the height above the bed at which the device is typically operated, the boundary Reynolds stress in the deflected outer part of the jet are nearly neutral. The flow can thus propagate across the bed without significant interaction with the overlying ambient fluid. This is in contrast to the wall jet flow created by impingement of a normal turbulent round jet.

The whole process is, therefore, highly efficient and can achieve a very high rate of excavation and sediment movement. However, while maximum excavation occurs at a Jet Swirl Number of 4, this is not necessarily the ideal Swirl Number for optimum long-distance transport. This is because, by the time the outer flow reaches the bed, its velocity is significantly attenuated. A lower Jet Swirl Number, giving a less splayed jet, may thus be more beneficial in terms of onward transport, particularly of coarser grained (sand) sediment.

In addition, the device can be used to excavate virtually any soil material, including very stiff clays, which are otherwise very difficult to excavate using conventional low-pressure water-jetting techniques. This enhanced excavation capability derives from the fact that in the presence of cohesive (i.e. low friction) bed materials a strong suction develops as the nozzle is brought closer to the bed. Suction results from the fact that in the rising swirling counter-flow stream, pressure decay occurs on the swirl axis due to strong flow convergence (c.f. a grounded tornado). Most cohesive soil materials are weakest in tension, compared to compression and shear, so this method of excavation, in contrast to normal jetting techniques, exploits the inherent weakness of the bed material.

Note that the aforementioned process also provides an effective means for cleaning, as discussed more fully in WO2004/0457750.

In the case of combustion applications, the important characteristics of the jet are: high turbulence, to create good fuel/air mixing; flame stability/length, and ingestion of hot gases to pre-heat the fuel/air mixture. The flame has to stay lighted and stable and so the appropriate length of flame depends to some extent on whether it is operating in a cross flow. To increase turbulence (for mixing purposes), advance gas

turbine combustors often use co-axial counter-swirling jets. Ingestion of hot gas is achieved by the axial counter-flow effect.

Notwithstanding the complex requirements associated with different types of combustion application, the ability to be able to pre-determine and control the characteristics of the jet by means of the flared nozzle is seen as being highly relevant to the field of combustion technology. Also of relevance, in terms of flame stability, is the ability to partition the flame into an outer zone with high velocity flow and an inner zone with strong re-circulating flow. While this partitioning occurs naturally in jets with high Jet Swirl Numbers, the effect is enhanced by means of the annular flow swirl generator. The annular flow swirl generator (component B1) also allows a much shorter flared nozzle to be used and a simpler overall design, by removing the need for a dual inlet swirl flow arrangement, which is currently the norm with counter-swirl combustors. Note that counter-swirl combustors are used where high rates of shear and mixing are required (see for instance: *Shin 1993*).

The particular design featured here (see Figure 19), uses a simple mono-bloc swirl generator (component B1) and flared nozzle (component A) construction, which is essentially the same as that shown in Figures 2 and 16, except that a tube 36 is inserted co-axially within the duct. Tube 57 provides the means for introducing a fuel stream into the jet at the point where the flared nozzle starts. Note this tube has the effect of making the duct swirling flow truly annular. The fuel stream would typically be a combustible gas/air mixture, pre-mixed in stoichiometric proportions, or a liquid fuel. The annular swirling flow, introduced through holes 6, would be a high-velocity air stream. The combustor is designed to operate at a Jet Swirl Number of between 2 and 4, to give a jet shape like that shown in Figure 20. Note that the higher the Jet Swirl Number the more the jet splays out. A high Jet Swirl Number is used to take advantage of the increased axial counter-flow 51, which reaches a maximum strength relative to the inlet flow at a Jet Swirl Number of 4.

Details of the near-field portion of the jet (flame) are shown in Figure 20. The fuel stream 58 is introduced at a velocity slightly less than the velocity of the counter-flow 51 so the fuel is immediately carried outwards along the bowl-shaped separation surface 59 between the outer high-velocity flow 53 and the counter-flow 51 and re-circulating flow 54. Note this separation zone is characterised by very high shear and turbulence and, therefore, high mixing rates. It is, therefore, an ideal location for combustion to take place. Because the flame is shielded, by the outer high-velocity air stream there is less risk of flame-out occurring due to external gusting effects. The velocity of this outer air stream would be matched to the flame combustion speed so the flame front is located in a ring-shaped zone 60 above the lip of the flared nozzle.

Because the counter-flow stream 51 brings hot combustion products back towards the flared nozzle the fuel is pre-heated prior to combustion, which is an advantageous feature. At the same time, the outwardly deflected counter-flow stream is cooled by the fuel, causing a radial outward increase in density. Since the outwardly deflected counter-flow stream is rotating due the effect of high shear between this stream and

the high-velocity swirling flow, the increase in density causes a strong centrifuging effect. This centrifuging effect helps to drive the secondary re-circulation and counter-flow. In the case of liquid fuel, the increase in density and centrifuging effect is caused not just by the cooling, but by the fuel droplets themselves. This is why it is important for the fuel to burn in a ring away from the axis.

While maintaining the general flame form, shown in Figure 20, the combustor would be tuned to provide optimum heat output and combustion efficiency. For spray applications, whether for spray drying or spray painting, a less splayed jet would be more appropriate, so a lower Jet Swirl Number (closer to 2) would be selected. Also a narrower high-velocity outer air stream would be used. Otherwise essentially similar design and performance criteria would apply. An advantage, of the design shown in Figure 19 as a spray-nozzle, is that only the co-axial tube comes in contact with the spray fluid. This means that it is easier to keep the nozzle clean. To clean out the co-axial tube, high-pressure air can simply be diverted through this tube.

As has already been mentioned, above a Jet Swirl Number of 1.5 a flow will break naturally as it emerges from a duct, without any additional help from an expansion nozzle. However, a flared nozzle (component A) of the present design may still be useful in order to control the breakdown process. Without such a nozzle, highly swirled flows issuing from a duct (such as from cyclone separators) can be very unstable and develop pronounced out-of-balance effects. This can lead to vibration and noise (see, for instance: *Yazdabadi P. and Griffiths A.J. Investigation into the precessing vortex core phenomenon in cyclone dust separators. Proc. Instn. Mech. Engrs. Vol. 208, 1993, pp 147-154*). The counter-flow stream may also penetrate and partially block the duct entrance, leading to loss of efficiency (see, for instance: *Griffiths, A.J., Yazdabadi, P.A. and Syred, N. Alternate Eddy Shedding Set Up by the Nonaxisymmetric Recirculation Zone at the Exhaust of a Cyclone Dust Separator. Journal of Fluids Engineering, Vol. 120, March 1998, pp 193-199*). In this instance, a longer flared nozzle may be selected, which is specifically designed to under-expand the flow. The jet thus experiences a lesser degree of breakdown than would occur without the flared nozzle. Similarly, the flared nozzle may be incorporated into pipework to control vibration and noise (in-duct application).

A further duct-exit application is in the cold outlet end of Ranque-Hilsch vortex tubes. Ranque-Hilsch devices (for a detailed working explanation see, for instance: *Cockerill, T.T. Thermodynamics and fluid mechanics of a Ranque-Hilsch vorted tube. MSc Thesis, University of Cambridge, Engineering Department, 1998.*), use a swirling flow, driven by compressed air, to effect a cooling at one end and a heating at the other end, of a vortex tube. In conventional apparatus, the swirling flow emitted from the cold end is initially allowed to expand, either through the centre hole in an orifice plate into a short length of tube, or through a conventional conical diffuser. Note the area expansion ration (AER) at the cold end (based on the hole diameter in the orifice plate and the diameter of the tube, is typically 4 (see www.visi.com/~darus/hilsch for a dimensional sketch of the apparatus). It, therefore, comes within the expansion capability of the present (flared nozzle) invention. Because of the highly swirled nature of the flow emitted from the cold end, the

elliptical flared nozzle (component A) described herein would provide a physically smaller and thermo-dynamically more efficient method of expanding the flow, compared to the longer conical diffuser or straight length of pipe. It also achieves superior mixing with the ambient air due to the re-circulating flow pattern developed.

It is anticipated that the elliptical flared nozzle principle, working in reverse, may also be used to more efficiently compress the gas during its transit to the hot end of the tube. Details of an apparatus, combining the expanding flared nozzle (component A), a passive annular swirl generator (component B1) and the elliptical flow contraction concept, are shown in Figures 21, 22 and 23. A flow compressing elliptical profile 61 is shown, forming part of the inner diameter of the hot tube 62, in Figures 21 and 23. Note this profile is confined to the distal (hot) end of the tube, which, over most of its length, has a uniform diameter. A simple thin-walled tube 63 is also used to physically separate the annular swirling flow 64, generated in the swirl chamber 7 (in component B1, a detail of which is shown in Figure 22), from the axial counter-flow 51 travelling towards the cold end.

The principle of operation of the present apparatus, based on the known general principle of operation of the Ranque-Hilsch vortex tube, is described, for completeness sake, as follows. For clarity of understanding, a detail of the flow at the hot end is shown in Figure 23. Compressed air enters inlet 65 and circulates around recessed chamber 66, before entering tangential slits 67 in annular swirl generator (component B1), a sectional detail of which is shown in Figure 22. Note annular swirl generator (component B1) would be designed to create an annular swirling flow with a Swirl Number of about 4, this being the optimum, in terms of strength of axial counter-flow development. Annular swirling flow 64 travels towards the hot end along the periphery of the tube, and as its diameter is reduced by the converging elliptical profile 61 the gas is compressed and gets progressively hotter. Compression of the gas results in a reduction (or non-increase) in longitudinal velocity in spite of the contraction in area. The Swirl Number of the flow also reduces due, in part, to the no-slip boundary condition, which reduces the swirl along the wall. In the face of the adverse pressure gradient, the gas also begins to separate along a wall-parallel surface 68 within the flow, with near-wall low-swirl hot flow 69 separated from high-swirl cooler axial flow 70. The separation surface 68 can thus be seen as the primary location where heat transfer takes place; heat transfer being a consequence, or associated feature, of vorticity being diffused away from the wall.

On the axis at the hot end, a bipolar jet develops, with virtually swirl-free hot gas exiting through a small circular opening 71 into a conical diffuser 72, and highly swirled cooler gas forming an axial counter-flow jet 51, which travels back towards the cold end. As this axial counter-flow travels along the tube it changes from a consolidated jet (see Figure 5) to a more annular jet (see Figure 4). As a result of this jet spreading, some re-circulation of axial counter-flow is envisaged to take place back along the tube, as indicated by the re-curved arrows in Figure 21. This is thought to be where secondary heat transfer takes place, with slightly warmer gas at the periphery of the axial jet being re-circulated back along the outer part of the tube to the hot end.

By the time the axial counter-flow stream 51 reaches the cold end it is amenable to rapid expansion through flared nozzle (component A).

The elliptical compression profile 61 of the hot tube can be seen as providing a more efficient flow-shaping, than typically occurs in a conventional fixed diameter tube, where a corner eddy is thought to provides a similar natural shaping effect. Note the corner eddy (ring vortex) is equivalent to the steady peripheral disturbance, as previously described. Because of the greater rate of compression and higher exit temperature as a result of the elliptical profile, a smaller inlet volume of gas is required to achieve the same temperature differential.

As an added refinement, it can be seen that honing a very smooth surface (very low friction) on the hot tube along its constant diameter section, while creating an etched surface (higher friction) on the elliptical profiled section, would further increase efficiency. This added efficiency (compared to existing apparatus) would be realised by a reduction in swirl (vorticity) content of the hot exit stream, to an absolute minimum, thus conserving vorticity within the apparatus; vorticity only being ejected at the cold end. Note that in conventional Ranque-Hilsch tubes the mass flow ratio of emitted hot and cold gas is typically about 3:1, respectively. By conserving vorticity this ratio can be reduced, thus reducing the total volume of gas required for a given temperature differential.

Note that since a flow-on-flow separation surface does not constitute a no-slip boundary, the corner eddy shaping effect would not result in conservation of swirl (vorticity) in the same way as the roughened elliptical profiling of the tube.

The optimum elliptical compression profile is designed on the basis of:

1. The difference in Swirl Number between the annular flow emerging from the swirl chamber (component B1) and that exiting at the hot end, and;
2. The relative diameters of the hot tube at its maximum diameter and the diameter of the exit hole.

The exit hole 71 at the hot end, clearly, has to be of sufficient size for compressed gas to transport heat out of the apparatus. There is, therefore, also an optimum ratio between the volume of gas (at standard temperature and pressure) exiting the hot end and the cold end. This means that the overall design has to be balanced (tuned) to achieve optimum performance. Further adjustment and tuning can be achieved by means of needle valve 73.

APPENDIX

The following definitions and descriptions are complied from standard texts:

Swirl Number

Swirl Number is a non-dimensional, device-independent measure of the ratio of axial fluxes of swirl and linear (axial) momentum in a flow, divided by a characteristic radius.

Swirl Number (S), is defined as:

$$S = \frac{1}{R}\frac{\int_{0}^{R} r^2 UV\,dr}{\int_0^R rU^2\,dr}$$

Where: R = inlet radius
r = radius of measurement
U = axial velocity
V = tangential velocity (swirl)

Reynolds Number

Reynolds number is a non-dimensional, device-independent measure of the ratio of inertial and viscous forces in a flow, multiplied by a characteristic diameter.

Reynolds number (Re), is defined as:

$$Re = \frac{\rho D u}{\mu}$$

Where: D = inlet diameter
ρ = fluid density
u = mean axial velocity
μ = fluid viscosity

Reynolds Boundary Stress

Reynolds boundary stress is a shear stress that occurs along the boundary between two fluids (typically one moving and the other stationary) which results in a momentum transfer between the two fluids. The stress is a result of the time averaged, fluctuating, components of velocity (axial and radial) within the flow, which may be positive or negative. If these two components are in-phase, the boundary stress is negative, if they are out-of-phase the boundary stress is positive. A negative boundary stress results in an inward transfer of momentum into the moving fluid (entrainment); a positive boundary stress results in an outward transfer of momentum from the moving fluid (detrainment).

Reynolds boundary stress (τ_R), is defined as:

$$\tau_R = \underline{\text{Force}} = -\rho u'v'$$

Area

Where: τ_R, u' and v' are time-averaged components
ρ = fluid density
u' = fluctuating component of axial velocity
v' = fluctuating component of radial velocity

Vortex Flows and Vortex Jet Core

The primary swirling flows to which the present specification refers can also be referred to as vortex flows. Vortex flows can be grouped into either forced-vortex flows: in which the tangential (swirl) velocity is zero at the axis and a maximum at the periphery; or free-vortex flows: in which the peripheral velocity is zero and the axial velocity is theoretically infinite.

Because most mechanical swirl generators are essentially forced-vortex generators, a ducted swirling flow from such a device (as described herein) will tend to have an outer part, which behaves as a free-vortex, and an inner part, which behaves as a forced-vortex.

The core is essentially the forced-vortex part of the flow and represents the diameter at which the tangential velocity is a maximum. In the case of a conventional propeller-generated swirling flow, the core often comprises by the axial hub vortex, which represents the combined (rolled-up) viscous boundary layer shed from the propeller blade surfaces. Because longitudinal flow converges towards this axis the core develops jet-like qualities.